Experimental evaluation of a heating radiant wall coupled to a ground source heat pump

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Abstract

A radiant wall heating system embedded in a heavy brickwork envelope and coupled to a ground source heat pump supplied has been experimentally tested under real outdoor conditions. This system was applied to a room sized cubicle built in Puigverd de Lleida (Spain) test-site, where it was studied in system vs. system analysis in comparison to a reference cubicle built with commercial available technologies (insulated alveolar brick wall and air-to-air heat pump). The results showed the potential of the radiant wall, which in continuous operation reached energy savings between 19.97 % and 40.72 % based on set-point temperature. Most important, the active thermal mass of radiant wall allowed operating in off peak periods. Otherwise, this peak load shifting ability was completely inexistent in the reference cubicle. However, the results show that the radiant cubicle was unsuited to operate in occupancy schedules due to its slow response time. Furthermore, the tests show that optimization of the radiant wall system requires a control strategy that takes in account the dynamics of the system.

Keywords: Thermally Activated Building Systems (TABS); Radiant Walls; Radiant Heating; Ground Source Heat Pump (GSHP)

1 Nomenclature

TABS	Thermally Activated Building Systems
VTABS	Vertical Thermally Activated Building Systems
TB	Thermal Barrier
HP	Heat Pump
GSHP	Ground Source Heat Pump
AAHP	Air to Air Heat Pump
COP	Coefficient of Performance
Top Geo	Operative temperature of geothermal cubicle
Top Ref	Operative temperature of reference cubicle

Tout	Outdoor ambient temperature
UBB	Unknown But Bounded
GSC	Gain Scheduling Control
PWM	Pulse Width Modulation
MPC	Model Predictive Control

2 Introduction

In the last decades, there has been much concern on the energy consumption and greenhouse gases emissions, which led to COP21 agreement [1] establishing the objective of maintaining the global average temperature increase below 2ºC above pre-industrial levels, but pursuing efforts to not reach an increase higher than 1.5ºC. Related to this point, on 2012 the International Energy Agency (IEA) [2] calculated that buildings account for 32 % of global energy use and almost 10 % of total direct energy-related CO_2 emissions. If electricity generation and district heating emissions are taken into account, then buildings are responsible for over 30 % of total end-use energy-related CO_2 emissions. Consequently legislations and regulations were issued to tackle this problem, as an example the European Directive 2010/31/EU [3] stands that by 2020 new buildings must be designed to consume "nearly zero" energy and existing buildings must be refurbished into very low energy buildings.

A promising technology for energy reduction in buildings is the thermally activated building systems (TABS). TABS are pipes or ducts embedded in the building surfaces or structures to work as heat exchangers, which provide heating and/or cooling to the rooms and store heat in the thermal mass of building components. These activated surfaces mainly exchange heat with the room in form of radiation, that means that TABS also directly exchange heat with the occupants. This characteristic implies both comfort and energy advantages. On the comfort side, heating and cooling with TABS reduces the required air change rate to the minimum for ventilation. Consequently, this reduces draught and noises. On the energy side, TABS can achieve operative temperatures inside the comfort range with variable air temperature, consequently reducing ventilation energy loses as indoor air has a temperature nearer to outdoor air temperature. However, the main point is that TABS make use of the buildings large surfaces, hence they can achieve high heat flux even with low temperature gradients. As a result TABS allow operation with high temperature cooling and low temperature heating. Furthermore, this low temperature gradient improves chillers and boilers performance, moreover, it makes the use of low grade energy sources feasible. Finally, TABS make and active use of the building thermal mass, which can be used for peak load shifting thus profiting of low cost energy periods and reinforcing the use of environmental energy sources available in short periods [4].

The benefits of peak load shifting have been studied for different TABS applications. In order to minimize cooling with a heat pump, Meierhans studied radiant cooling applied to an office building [5,6]. The ceiling slabs shifted the cooling load to night-time when free-cooling was available as outdoor cool air. The monitoring of a room showed that the system could maintain comfort temperature even with internal and solar heat gains. Similar conclusions were obtained by Olesen et al. [7], whose dynamic simulations pointed that TABS could help in reducing and shifting operation time by night-time activation or intermittent operation of the supply pumps, all without losing comfort. On that point, Bauman et al. [8] compared the operation schedules of monitored radiant slab office buildings. First, using a schedule typical to quick response all-air system and then contrasting it with a night-time pre-cooling schedule. The comparison showed that the radiant slab could successfully maintain comfort conditions. Furthermore, comfort surveys showed a good occupant satisfaction. The benefits of peak load shifting extend also to supply systems. On this point dynamic simulations by Dossi et al. [9] showed that night-time heat storage helped reducing peak load, which improved heat-pumps COP. Furthermore, TABS were also useful to reduce operation cost, as operation could be shifted to low cost periods. On that point, Sarbu and Sebarchievici [10] showed that a radiant floor required less on/off switching than a radiator, which reduces the stress on the heat pump equipment, thus it required less maintenance and it had a longer life time.

The TABS high thermal mass that makes possible to shift peak loads also makes their control challenging. In that sense, heating curve and indoor temperature feedback have been standard in TABS control strategies. However, control strategies have been improved with better definition of heating and cooling curves in form of Unknown But Bounded method (UBB) [11,12] and techniques to define the active periods such as the improvement of PID through Gain Scheduling Control (GSC) [13,14], the calculation of the heat pulse with Pulse Width Modulation (PWM) [15,16], the use of statistically identified model in Model Predictive Control (MPC) [17,18] or with the development of adaptive and predictive controls [19,20].

Most of TABS that have been researched and applied to buildings were placed on horizontal surfaces, such as floors, ceiling, and in-floor slabs. However, TABS can be also placed in vertical surfaces with panels or embedded into walls. In one of these cases, Zhu et al. [21] studied pipe-embedded envelopes. This type of TABS is placed on outer walls and in that specific case the supplied water was used for intercepting the heat waves from outdoor ambient and, consequently, reducing cooling and heating demand. The heat sources and sinks were cooling towers and ground heat exchangers for cooling or solar collectors and ground heat storage for heating. The results of the study showed that pipe-embedded envelopes actively intercepted heat/cold and reduced the cooling/heating load of the building. The model developed in this study, a 2D model Frequency Domain Finite Difference (FDFD) was further used for a parametric study [22]. Later, the model was improved with genetic algorithm for defining the model parameter [23], afterwards it was coupled to the Number of Transfer Units (NTU) method for considering the temperature variation in the pipe direction [24]. Krzaczek and Kowalczuk [13] applied the same concept, in this case called Thermal Barriers (TB), but coupled to heat storage system consisting of ideal ground heat sources divided in three levels: low temperature, medium temperature, and high temperature heated with solar collectors. The results also showed the capacity of the TB for reducing the heating and cooling demand of the building. In this study a Finite Elements Model (FEM) was developed and used for a parametric study of the TB. The research continued with the development of a Fuzzy Mixing Gain Scheduling (FMGS) controller for TB [14]. Complementary, the components and structure of the TB were studied [25]. Also in a simulation study, Bojic [26] used EnergyPlus to show the better synergy of radiant panels with the building insulation compared to conventional radiators. Aside from simulation research, Venko et al. [27] made a laboratory experimental study. The focus was on the convective part of the radiant panels, studying the activation length of vertical TABS on mixed convection conditions.

As previously stated, the use of TABS can enhance the use of renewables. In this context low enthalpy geothermal systems combined with heat pumps have been commonly coupled to TABS. Ground Source Heat Pumps (GSHP), also called Geothermal Heat Pumps, are systems combining heat pumps with a ground heat exchanger (closed loop systems) or fed by groundwater from a well (open loop systems) [28]. These systems use the constant temperature of the ground, whose thermal capacitance reduces the daily and seasonal temperature fluctuation as the depth increases [29]. The ground temperature stays around the yearly average outdoor temperature, consequently being higher than the average outdoor temperature in winter and lower in summer. As a result, using the ground as heat source or heat sink increases the efficiency of heat pumps (HP) both in heating and cooling modes. The performance of HP can be further improved with TABS because they require a supply temperature lower for heating or higher for cooling, therefore GSHP coupled to TABS benefit from a reduced temperature difference between the evaporator and the condenser.

Despite many research has been conducted both on TABS and GSHP, there are no examples that integrate both a VTABS and GSHP in pilot plan experimentation. Furthermore, most VTABS research has been done in simulations works or at laboratory tests. Consequently, testing of VTABS under real outdoor conditions is valuable for obtaining information to evaluate the potential of such systems and to validate simulation models with real data.

In this context, the present paper describes the experimentation under real outdoor conditions in a house-like prototype scale. The study was based on system vs. system comparison, the cubicles built allowed for a comparative study of buildings with equivalent envelopes but using different HVAC systems and distribution system, in that case a radiant wall coupled to a GSHP on a side and a conventional air-to-air heat pump on the other. The studied radiant wall differs from other VTABS found on the literature, as most radiant systems are embedded into concrete [13,23] or made of radiant panels [26,27] while the cubicle built for this study had the radiant system embedded into a heavy brick wall. The present experimental study pretends to demonstrate the energy savings potential of the system as well as its peak load shifting ability.

3 Experimental set-up

The experimental set-up consists of three house-like cubicles built in Puigverd de Lleida (Spain) experimental site. There are two reference cubicles and a study cubicle, all shown in Figure 1. The internal size of the cubicles is 5.25x2.7x2.7 m.

The radiant wall cubicle is built based on 185 mm wide alveolar bricks. Polyethylene pipes (18 mm diameter) are embedded at 36 mm depth, in grooves built on the internal surface of the wall with 150 mm spacing. On the external part of the envelope there are 60 mm of expanded polystyrene insulation and an open joint ventilated facade. The radiant walls are divided into five loops (two on the south and one on the other walls) which are connected to a common manifold. The manifold receives and distributes the water supplied by an ecoGEO B2 GSHP [30], which has a nominal heating power of 1.4 kW and a coefficient of performance (COP) of 4.6 according to EN 14511 [31]. The heat pump supplies heat according to the heating curve shown in Figure 2. The ground source system consists of two boreholes, both containing two U-pipes of different lengths, 20 m and 40 m each.

Figure 1: Radiant Wall cubicle and reference cubicle (left) and detail on installation of embedded pipes (right)

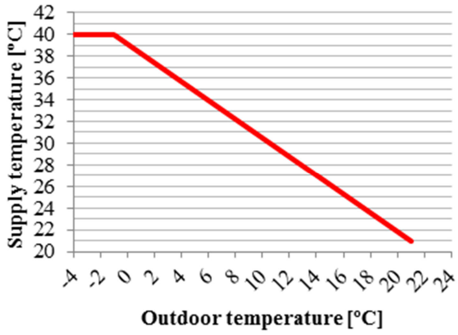
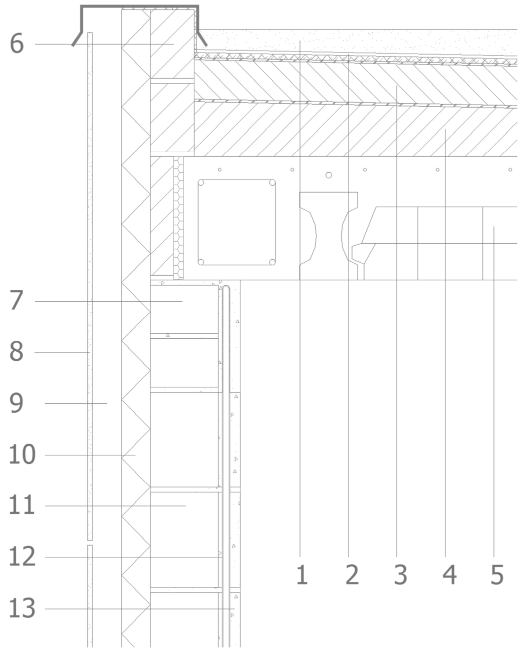


Figure 2: GSHP heating curve

To compare the radiant wall cubicle performance against a conventional system, two reference cubicles were built based on the same alveolar bricks constructive system and equal envelope thermal transmittance in steady-state (U value of 0.5). The roof was built with the same systems as the radiant cubicle. Therefore, the only differences between the radiant and reference cubicles are the heating and cooling system and the external ventilated facade. The reference cubicles are equipped with Fujitsu ASHA07LCC air-to-air heat pumps (AAHP) which have a nominal power of 3 kW and a nominal heating COP of 4.55. As there are no difference between reference cubicles, which were both built alike, only the results of one will be reported in this paper except for enhanced operation test, when each reference cubicle had a different set-point.

More details about the cubicles construction can be found in Romaní et al. [32].



1 - Gravel
2 - Double asphaltic membrane
3 - Polystyrene 8cm
4 - Cement mortar, flat roof 3% sloped
5 - Concrete precast beams + 5cm concrete slab
6 - Hollow brick
7 - Alveolar brick
8 - Ventilated facade panel
9 - Ventilated facade channel
10 - Polystyrene 6cm
11 - Alveolar brick
12 - Polyethylene pipes
13 - Groove for pipe filled with cement mortar

Figure 3: Wall and roof constructive details

The following parameters were measured and registered at 5 minutes interval:

- Internal surface temperature of walls, roof and floor (Pt-100 DIN B calibrated with a maximum error of ± 0.3 ºC).
- Borehole temperatures at 5, 10, 20, 30 and 40m (Sheathed Pt-100 DIN B calibrated with a maximum error of ± 0.3 ºC)
- Inside temperature and humidity (ELEKTRONIK EE21 with an accuracy of ± 2 %).
- External temperature and humidity (ELEKTRONIK EE21 with an accuracy of ± 2 %).
- Horizontal global solar irradiance (Middleton Solar pyranometer SK08 ± 2 $W \cdot m^{-2}$).
- Electric energy consumption (Circutor MK-30-LCD-RS485 with an accuracy of ± 1 %)
- Pulse flow meters (Zenner MTKD-N with 1 pulse per litre and maximum operative temperature of 50ºC)

4 Methodology

The current paper analyses the performance of the radiant wall coupled to a GSHP in heating mode. The experimental campaign was focused on the influence of the set-point temperature on energy use, the testing of different operation schedules, and the evaluation of the wall thermal storage capacity.

4.1 Tests

The following tests were carried out to evaluate the radiant wall features:

- Continuous operation: Heating was operated to maintain a constant set-point temperature during the whole experiment. The main goal of the test was to compare the electrical energy consumption in each cubicle at the set-point temperatures 22 ºC, 24 ºC, and 26 ºC, which are a set of temperatures that represent all the temperature range for comfort.
- Occupancy schedule operation: The cubicles were heated at a set-point of 24ºC only during occupancy schedules. Two different schedules were used: domestic schedule (from 17:00 to 8:00) and office schedule (from 8:00 to 13:00 and from 14:00 to 17:00). The purpose of these tests was to study the dynamics of the system, the potential to achieve comfort conditions, and the electrical energy consumption when operation time was limited to an occupancy schedule.
- Night-time charging: Heating was operated from 0:00 to 8:00 at set-point 26 ºC, thus pre-heating the cubicles when energy cost was lower. This experiment studied the peak load shifting capacity of the cubicles, showing its capacity to maintain the comfort conditions during the day while it was only active for a short period. The main parameter in this test was the time outside comfort range at each cubicle.
- Enhanced operation test: The objective of this test was to make each system maintain comfort conditions during the whole day taking advantage of their most adequate operation mode. Experience from previous tests showed that reference cubicles had to operate their AAHP as in continuous operation test, without varying its set-point. The set-points selected were 22ºC and 24ºC for each reference cubicle. On the other side, the radiant cubicle operated with a set-point of 26ºC during pre-heating period at night and a set-point of 22ºC for the rest of the day. The pre-heating period would store energy during low cost periods and reduce the energy demands for the rest of the day. The set-point of 22ºC during the rest of the time would guarantee that comfort conditions were maintained during the whole day. Two schedules were tested, one with pre-heating from 0:00 to 8:00 and another from 6:00 to 8:00.

4.2 Measurements

Measurements were taken every 5 min thus instant power could not be represented. This was especially important for AAHP, as figures would never show their nominal power because

AAHP only worked for fractions of time inside the measured time step, thus the measurements would only show average power during that period. Consequently, the chosen option was to represent hourly electrical energy consumption. Contrary, the thermal energy of the radiant walls and the boreholes was calculated assuming that the fluid temperature measured every 5 min could applied to the whole time step. In that case the flow was calculated as an average of the number of litres registered inside the time step.

Operative temperatures were used to assess the thermal conditions inside the cubicles. This gave a better indicator of the behaviour of the radiant wall cubicle. Calculations of the operatives temperatures were done according to chapter 8 of ASHRAE Handbook Fundamentals [33] for a point at the centre of the cubicle at a height of 1 m. Note that air temperature sensors were placed at the mentioned position for an accurate calculation of the operative temperature. It was assumed that air speed was low so that operative temperature could be considered as the average between mean radiant temperature and air temperature.

4.3 Error propagation

The two main parameters used in this study were the electrical energy use and the operative temperature.

The calculation of total and hourly electrical energy was simply the amount of the energy measured in a period, the whole test and one hour respectively. In this case, the error was simply the sensor measurement error (1 %).

In contrast, the operative temperature was calculated by taking in account the indoor air temperature and the surface temperature of the walls, floor and ceiling. The error propagation for the operative temperature was calculated with the general functions method [34] with an average error of 0.2 ºC in all measurements.

4.4 Comfort evaluation

In terms of comfort, the acceptable thermal range considered was 21 - 25.5 ºC. It was considered that potential occupants could adapt their clothing, e.g. by putting on or removing their warm clothes, thus the thermal ranges comfort obtained with clothing values of ~1 and ~0.5 considered in ISO 15251 [35] were stacked. In order to maximise the comfort, only category I comfort ranges for offices or residential building were used.

Complementary, the methodology of Hauser et al. [36] was used for characterising the level of comfort in night-time charging tests. This method considers both the time outside comfort limits and the temperature difference from this comfort limit. The parameter used is the excess temperature degree-hours coefficient, which was calculated with equations (1), (2), and (3). These equations were adapted to consider both the upper and lower limit proposed previously.

$$Gt_{Excess\ Temperature\ High} = \sum\left(T_{op} - 25.5°C\right) \cdot t \ \text{ for } \ T_{op} > 25.5°C \qquad (1)$$

$$Gt_{Excess\ Temperature\ Low} = \sum\left(21°C - T_{op}\right) \cdot t \ \text{ for } \ T_{op} < 21°C \qquad (2)$$

$$Gt_{Excess\ Temperature\ Total} = Gt_{Excess\ Temperature\ High} + Gt_{Excess\ Temperature\ Low} \qquad (3)$$

5 Results and discussion

5.1 Continuous operation test

The test at set-point temperature 22ºC was carried out from November 5th to 12th. Moreover, in the case of cubicles operating with a set-point of 24ºC, the tests were performed from December 5th to 12th. Finally, the test at set-point 26ºC took place from January 8th to 15th. Table 1 and Figure 4 summarise test conditions for all continuous operation tests.

Table 1: Test conditions for continuous operation test

	Average outdoor temperature (ºC)	Average minimum outdoor temperature (ºC)	Average maximum outdoor temperature (ºC)	Average accumulative daily radiation ($MJ \cdot m^{-2}$)	Accumulative daily radiation standard deviation
22ºC	13.44	8.64	19.83	7.64	4.98
24ºC	7.98	1.85	15.17	5.07	1.80
26ºC	9.67	2.99	16.00	7.34	1.52

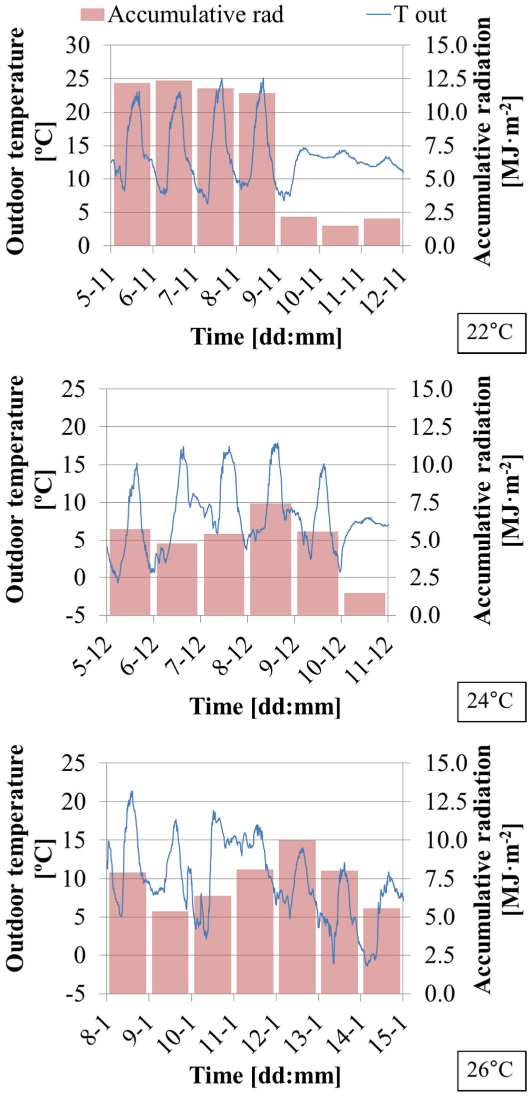


Figure 4: Outdoor temperature and daily accumulative radiation during continuous operation test

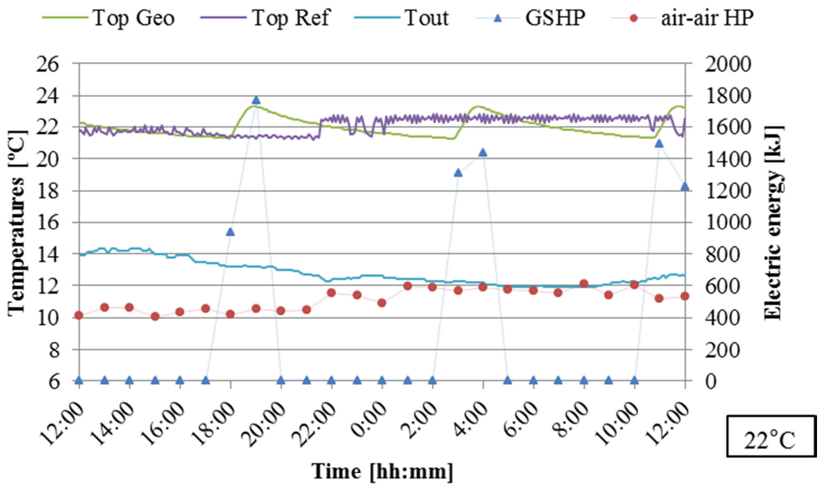
The radiant wall cubicle used less energy than the reference at all the analysed set-point temperatures, as summarized in Table 2. These savings depended strongly on the set-point temperature, as the advantage of the radiant cubicle was reduced when the set-point temperature was increased. Moreover the COP values calculated at each set-point were lower than the heat pump nominal value of 4.6. Despite this, the COP was kept at around 4 during the operation conditions of the test, in which the inlet temperature to the evaporator was never lower than 12ºC and the outlet water temperature from the condenser was always below 40ºC. These COP values were slightly lower than those found in the literature. According to the performance chart presented by Kim et al. [37] the COP should be around 5 with similar conditions. Michopoulos et al. [38] presented seasonal performance study of GSHP in Greece in which the seasonal COP for the heating season started at 4.4 on the first year and it later increased up to 5.2 due to storage effects in the ground, which also resulted in a decrease of the performance in cooling seasons.

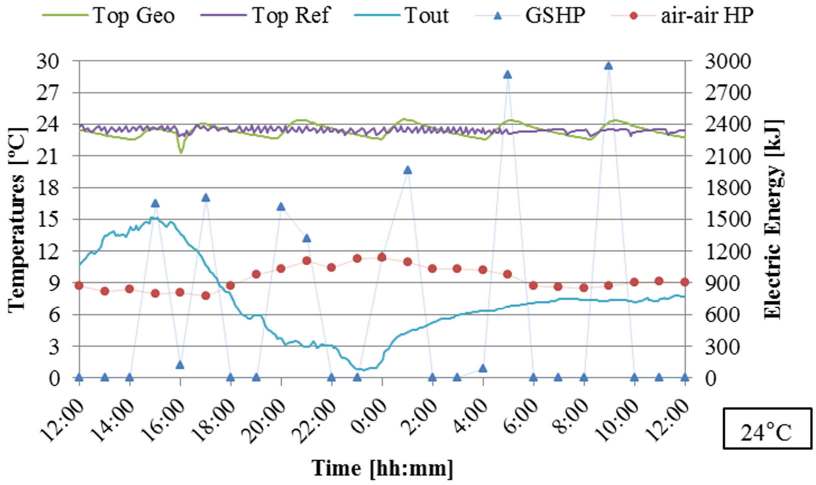
Table 2: Continuous operation test operation time, energy use and COP (according to test conditions in Table 1 and Figure 4)

		Radiant	Reference
22ºC	COP	4.27	-
	Energy (kJ)	49968	84322
	Savings	40.74 %	
24ºC	COP	3.96	-
	Energy (kJ)	112618	154432
	Savings	27.08 %	
26ºC	COP	4.02	-
	Energy (kJ)	168850	210985
	Savings (%)	19.97 %	

Figure 5 shows the operative temperatures and the electrical power consumption profiles for each cubicle. The AAHP was being activated during the whole day, however, as reflected by the average energy used the AAHP only worked at partial loads, thus being started and stopped in short cycles. On the other side, the GSHP had longer activation cycles, which implied longer periods without providing heating. This was caused by the controller dead-band, which allowed operative temperature fluctuation about 0.5ºC. The system used this dead-band to manage the slow response time of the radiant wall and thus avoiding overheating. It might also have helped to reduce energy use, but the obtained savings were higher than the ones that the use of dead-band would achieve on their own [39], especially in the tests with lower set-points.

In terms of energy use, the AAHP had the influence of outdoor air temperature as can be observed in the test with set-point 24ºC of Figure 5. In contrast the GSHP did not seem to be affected by outdoor temperature. The set-point temperatures also affected to the behaviour of both systems. On one hand, the increase of the set-point resulted in more activations of the GSHP, as shown on Figure 5, where there were 3, 6, and 7 activations respectively. On the other hand the AAHP had higher average energy use at higher set-points, rising from 500 kJ, to 950 kJ, and 1400 kJ per hour at set-points 22ºC, 24ºC and 26ºC respectively. Notice that energy use was not only dependent on the set point but on the heating demand, as well. In its turn, heating demand depends on the weather conditions reflected in Table 1 and Figure 4.





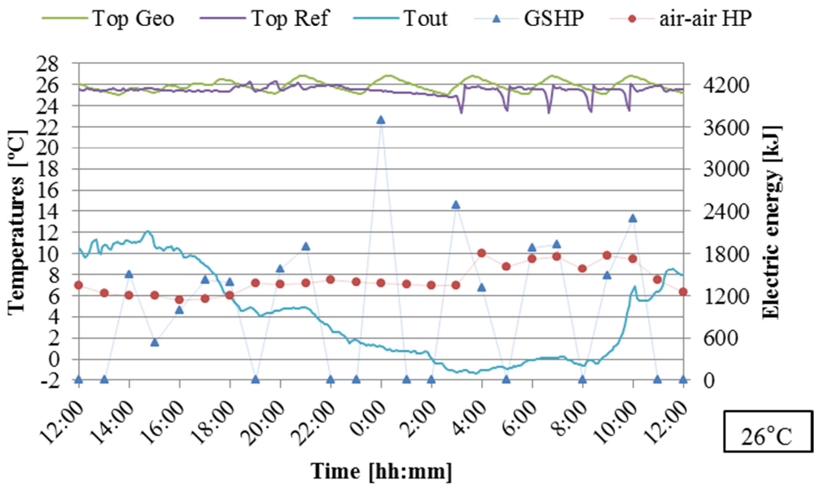


Figure 5: Hourly electric energy consumed and temperature profiles on continuous operation tests

5.2 *Occupancy schedule test*

The test with domestic occupancy schedule was carried out from November 28th to December 4th. Whereas, the test with office occupancy schedule was carried out from December 12th to 18th. Weather conditions for both tests are summarized in Table 3 and Figure 6.

Table 3: Test conditions for occupancy schedule tests, set-point 24ºC

	Average outdoor temperature (ºC)	Average outdoor minimum temperature (ºC)	Average outdoor maximum temperature (ºC)	Average accumulative daily radiation (MJ·m^{-2})	Accumulative daily radiation standard deviation
Domestic schedule	2.90	-3.05	12.03	7.80	2.60
Office schedule	4.50	1.70	9.63	3.75	1.95

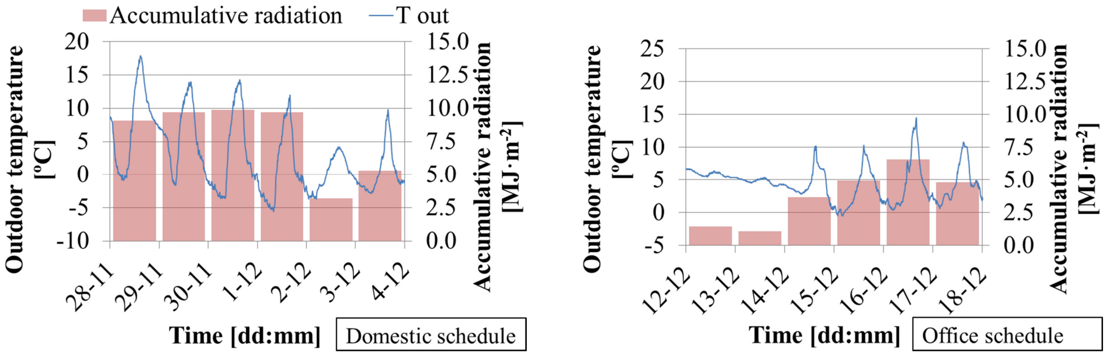


Figure 6: Outdoor temperature and daily accumulative radiation on occupancy schedule test

Operation according to occupancy schedules benefited the AAHP as it reduced the number of hours it was required for heating. However, in domestic schedule, the AAHP was forced to work when outdoor temperatures were lower, thus it worked during less efficient conditions and as a result it ended up consuming more energy than the GSHP. Despite this, the energy saving in this schedule was lower than the one achieved during tests under continuous operation at the same set-point (24ºC), as the reduction of operation hours is significant. In comparison, an office schedule had even a shorter operation time and the AAHP operated in more efficient conditions, when outdoor temperatures were higher, as shown on Figure 8. Consequently, in this case the AAHP consumed significantly less energy than the GSHP coupled to the radiant wall. The results of these tests are shown on Table 4.

A cause for these results was that the GSHP activated the thermal inertia of the building envelope, storing heat in the wall before heating the indoor space. Because of the thermal lag, part of the stored heat was released outside the activation period, thus it was wasted as it heated the cubicle during non-occupancy periods. Furthermore, at the beginning of the active period the GSHP required high power to compensate that the radiant wall had been left to cool down during non-active period to a low temperature. In contrast, the AAHP only heated the indoor air thus it needed less energy and obtained a faster response time. However, the AAHP achieved lower operative temperatures as it did not manage to store heat in the walls, as can be seen in the fast drop of temperature after the activation period shown on Figure 7 and Figure 8. The test showed that operation in these occupancy schedules was negative for the radiant wall as in these cases the thermal inertia was not exploited and ended up to be detrimental.

Table 4: Occupancy schedule test operation time, energy use and COP (according to test conditions in Table 3 and Figure 6)

	Domestic schedule test (24ºC)			Office schedule test (24ºC)		
	Energy (kJ)	COP	Savings (%)	Energy (kJ)	COP	Savings (%)
Radiant	152532	3.77	13.53 %	63982	3.92	-29.54
Reference	176389	-		85860	-	

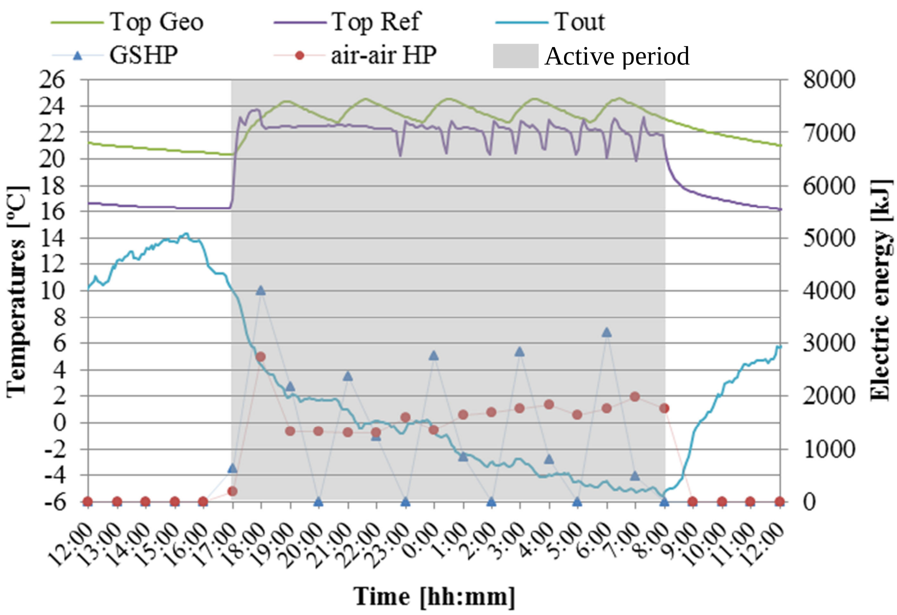


Figure 7: Hourly electric energy consumed and temperature profile on domestic schedule operation test

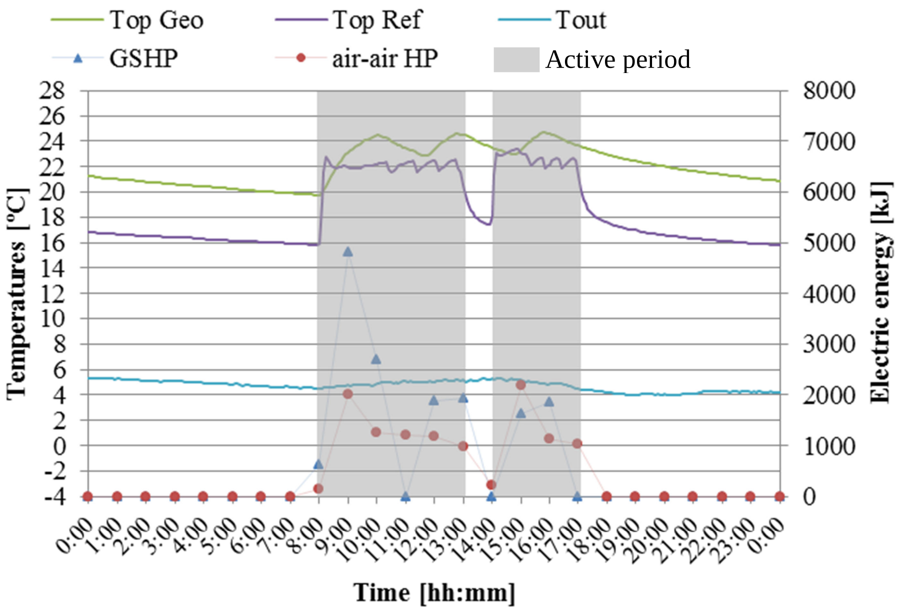


Figure 8: Hourly electric energy consumption and temperature profile on office schedule operation test

5.3 Night-time charging

The night-time charging test studies the capacity of the radiant wall cubicle to maintain comfort conditions in an off peak operation schedule. This test was carried out from December 19th to January 7th. In the tested conditions, with set-point of 26ºC during night-time period for both cubicles, the GSHP consumed 15 % more energy than the AAHP. However, a more relevant finding is that the radiant cubicle stays inside the comfort range most of time outside the active period, while the reference cubicle rapidly fall outside the comfort range.

As shown on Figure 9, the reference cubicle only stayed in comfort conditions during the activation period, that was the 34 % of the day, as shown on Table 5, where the comfort temperature bands are those between 21-23.5ºC and 23.5-25.5ºC Furthermore, during this period the AAHP did not manage to achieve the set-point temperature as it could not transfer heat to the wall efficiently, resulting in a lower operative temperature.

On the other hand, the radiant cubicle stayed inside the comfort range for more than 70 % of time. Additionally, it has to be noticed that the GSHP controller had to be set to 26ºC, then it exceed the upper limit of the comfort range (25.5ºC). As a result, part of the time outside the comfort range was due to overheating. Nevertheless the lower limit of comfort range was exceeded much less than in the case of the reference cubicle. Despite the radiant cubicle stayed

70 % of time inside the comfort range, it is important to highlight that its deviation from the comfort limits were below 1ºC, as Figure 9 shows. The deviation from comfort range is better reflected by the excess temperature degree-hours coefficient [36], here the advantage of the radiant cubicle was clear as it had a total excess degree-hours coefficient of 58.87 ºC·h against the 1700.19 ºC·h of the reference cubicle over 464 h of testing. As a comparison value, the excess degree-hours coefficient of ambient temperature was 6759 ºC·h. This shows that the radiant wall stayed more time inside the comfort limits and had smaller deviations once those were exceeded.

It is important to notice that the GSHP was activated for multiple times during the operation period, as can be observed on the multiple temperature spikes shown on the active periods in Figure 9. Thus, the system did not actually work in a true pre-heating mode but in a way such as if the occupancy period was during the night-time, when a set-point of 26ºC was required. The limitations of the controller resulted in the GSHP maintaining the set-point temperature for several hours instead of achieving the set-point temperature at the required time, which should have been the end of the pre-heating period which corresponds to the beginning of the occupancy period. A more efficient controller should predict the adequate activation period so the set-point temperature is achieved just before the occupancy period, consequently consuming less energy. Moreover, it could adjust the set-point, or more precisely the heat pulse, according to the weather forecast and expected internal gains. The literature already presents many examples of improved control strategies such a GSC [13,14], PWM [15,16], MPC [17,18], and adaptive and predictive controls [19,20].

Table 5: Time distribution of operative temperature ranges in night-charging test

Temperature range (ºC)	Radiant cubicle		Reference cubicle		
	h	%	h	%	
<21	19.75	12.65	102.5	65.7	
21-23.5	65.67	42.07	27.08	17.35	Comfort range
23.5-25.5	44.42	28.46	26.5	16.98	
>25.5	26.25	16.82	0.00	0.00	

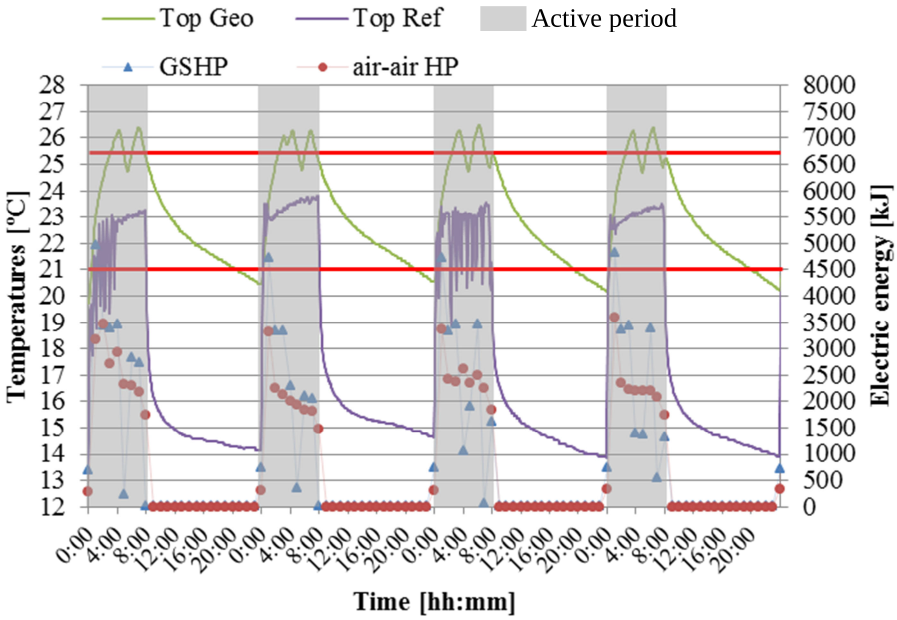


Figure 9: Hourly electric energy and temperature profiles on night-time charging test

5.4 Enhanced operation

The test contrasting the performance of the reference and the radiant cubicle in different operation modes were carried out in consecutive periods. Test with pre-heating from 0:00 to 8:00 was carried out from February 25th to March 3rd. On the other hand, the test with pre-heating from 6:00 to 8:00 was carried out from March 3rd to 11th. Test conditions for suitable test are described on Table 6 and Figure 10.

Table 6: Test conditions for enhanced operation tests

	Average outdoor temperature (ºC)	Average outdoor minimum temperature (ºC)	Average outdoor maximum temperature (ºC)	Average accumulative daily radiation (MJ·m^{-2})	Accumulative daily radiation standard deviation
Pre-heating 0:00 to 8:00	8.54	3.58	15.13	12.96	3.75
Pre-heating 6:00 to 8:00	8.52	2.9	15.33	17.51	4.13

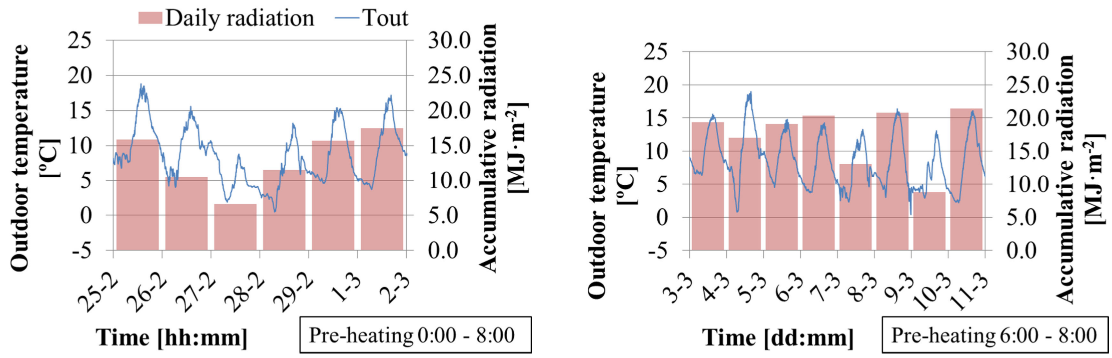


Figure 10: Outdoor temperature and daily accumulative radiation for enhanced operation tests

As summarized on Table 7, the radiant cubicle managed to save energy compared to both reference cubicles in the two pre-heating schedules. These savings were lower than in continuous operation test but the behaviour shown by the radiant cubicle gave interesting insight for control optimization. In most of the tested days, the same problem from night-time operation test was observed. The GSHP had to activate multiple times during the pre-heating in order to maintain the higher set-point, as Figure 11 shows. This suggested that the GSHP would have performed better if the pre-heating period had been shorter. Within this context, Table 7 shows that the pre-heating period from 6:00 to 8:00 resulted in higher savings. However, the shorter pre-heating caused that the GSHP had to activate during the rest of the day to avoid the indoor temperature dropping below the comfort limits. In that sense, Figure 12 shows a day with several peaks of activation outside the pre-heating period. Comparing the energy consumed by both schedules during the pre-heating period it was observed that it represented the 91.02 % of total energy use in the pre-heating scheduled from 0:00 to 8:00 while it supposed only 49.74 % in the schedule from 6:00 to 8:00, as shown on Table 8.

A main incentive for shifting operation to night-time periods can be found on variable energy cost. The current case considered variable tariff in Spain with an electricity cost of 0.145586 €·kWh^{-1} from 12:00 to 22:00 and 0.065514 €·kWh^{-1} the rest of the time [40]. However, the fix cost was not taken in account, as it was related to the power term and taxes among which little savings can be achieved outside district heating. According to these tariffs, the economic savings with pre-heating were significantly higher than the energy savings alone would suggest, as shown on Table 7. As expected, the difference between energy savings and economic savings were more important for the test with pre-heating from 0:00 to 8:00, as it concentrates its operation during the low cost period, as shown in Table 8, as opposite to the pre-heating from 6:00 to 8:00, whose activations outside the pre-heating period significantly increase the cost. Nonetheless, savings were still higher with the shorter pre-heating period.

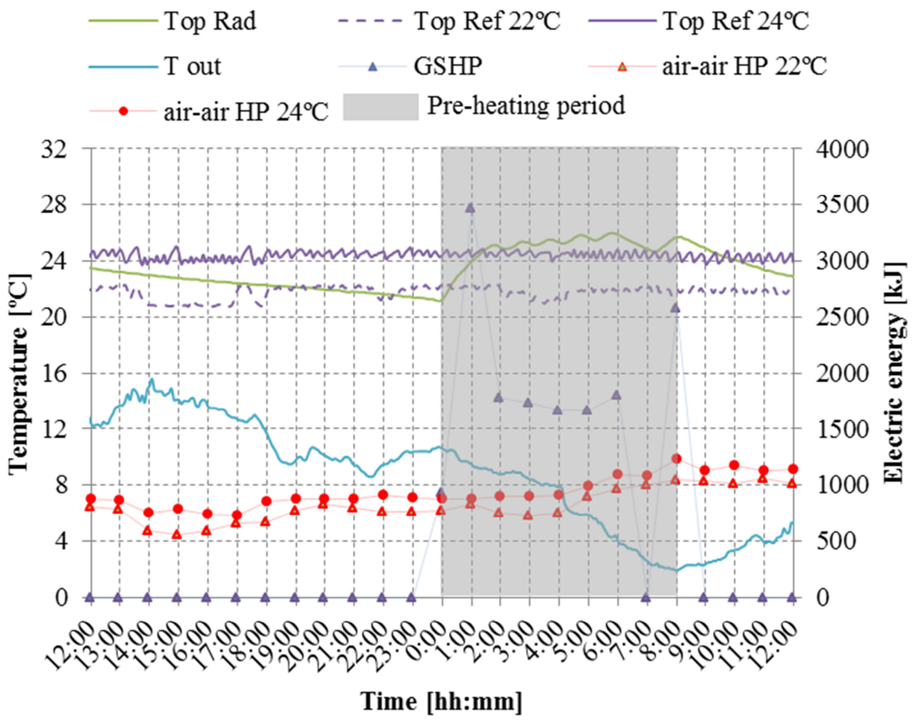


Figure 11: Enhanced operation with GSHP set-point of 26ºC from 0:00 to 8:00

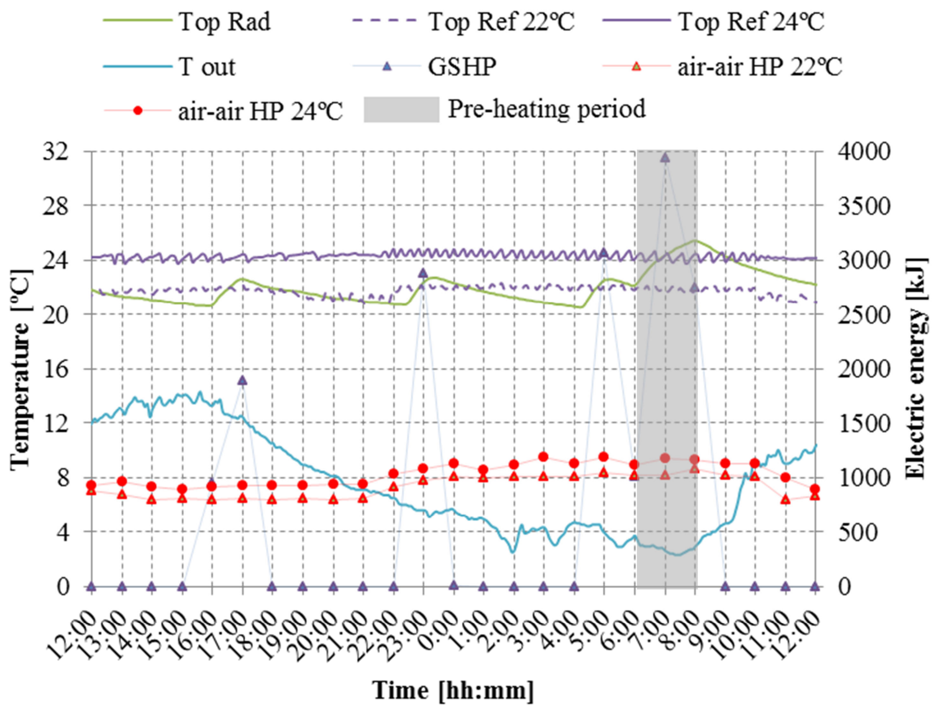


Figure 12: Enhanced operation with GSHP set-point of 26ºC from 6:00 to 8:00

Table 7: Enhanced operation results (according to test conditions in Table 6 and Figure 10)

	Reference		Radiant		
	Set-point	Energy (kJ)	Energy (kJ)	Energy savings (%)	Economic savings (%)
Pre-heating from 0:00 to 8:00	22ºC	135352	128451	5.10 %	31.54 %
	24ºC	152604		15.83 %	39.47 %
Pre-heating from 6:00 to 8:00	22ºC	179172	131432	26.64 %	38.63 %
	24ºC	203583		35.44 %	46.32 %

Table 8: Detail on energy use and energy cost of the radiant cubicle in the pre-heating period and rest of the day

		Energy use (kJ)		Energy cost (€)	
Pre-heating from 0:00 to 8:00	Pre-heating period	116917	91.02%	2.13	88.60%
	Rest of the day	11534	8.89%	0.27	11.40%
	Overall	128451		2.40	
Pre-heating from 6:00 to 8:00	Pre-heating period	65379	49.74%	1.19	41.90%
	Rest of the day	66052	50.26%	1.65	58.10%
	Overall	131432		2.84	

6 Discussion

The experimental campaign has shown the potential of the radiant wall coupled to a GSHP to save energy compared to a conventional system featuring a heavy brick wall and an AAHP. However, these savings were highly influenced by the set-point and the operation schedules. The maximum savings obtained were around 40% with set-point of 22ºC and in continuous operation, that is maintaining the set-point all the day. Although the set-point is in the comfort range it is rather low and it is possible that real applications might demand a higher set-point. Furthermore, continuous operation would not be a realistic operation strategy, especially for AAHP which are commonly operated only in occupancy schedules.

Regarding the control strategies, the different test showed the great influence that operation schedules have on TABS performance. At first, the thermal inertia of the radiant wall had a slow response time, which makes it ill-suited to operate under occupancy schedules, contrary to AAHP. In case of the office schedule, the thermal lag of the radiant wall caused that the set-point was achieved about 2 hours after the beginning of the occupancy. Furthermore, the temperature was maintained around the set-point for some hours after the occupancy period. That means that part of the heat supplied early in the morning was stored in the radiant wall and it was released outside the occupancy period, consequently, it can be considered that this heat was wasted in terms maintaining comfort conditions during the office schedule. However, while the thermal inertia did not allow operation on occupancy schedule it allowed shifting the peak loads. That meant that the system could store heat in an off-peak period, which might be more energy efficient or less expensive, and still maintain comfort during the occupancy periods. These results agreed with previous studies found in the literature, which already showed the potential of TABS for peak load shifting strategies [5-10].

Furthermore, the night-time charging test and enhanced operation test highlighted the importance of and optimized controller. The three tests clearly showed the potential of peak load shifting with TABS, as all stored heat during night period and kept good comfort conditions during the day. However, none of them were optimized. The pre-heating period from 0:00 to 8:00 was too long, and the system supplied excess heat as it kept the higher set-point for a long time. Furthermore, in the night- time charging test the set-point was much higher than what was probably needed. In contrast, the pre-heating period from 6:00 to 8:00 seemed to short in the tested conditions, as the cubicle required heating outside the pre-heating period much often and earlier. These results already show a need to adapt the heat pulse to a predicted heat load according to the thermal state of the building and the outdoor conditions. Furthermore, this is more relevant if the variations of heating load along the winter seasons are taken into account. That strongly suggests that PWM [15,16], MPC [17,18] or adaptive and predictive control [19,20] should be tested in the radiant wall cubicle. This will have a strong interest, as these controls have been applied to horizontal TABS, but not to VTABS exposed to real outdoor conditions. As pointed in previous research [13,14] VTABS on the envelope of the building also help in reducing the influence of outdoor conditions, asset that might add to better the comfort provided by radiant systems and its thermal storage capacity.

7 Conclusions

An experimental study is presented for the heating performance of a radiant wall cubicle coupled to a ground source heat pump. The objectives of the experimentation were to test the performance of the system compared to an equivalent reference cubicle. The study took in account different set-point and the evaluation of the peak load shifting ability, as well as considering insights for control optimization.

The experimental measurements demonstrate the potential of the analysed system to achieve energy savings in comparison to a standard air to air heat pump. At most of the test, the system used less energy than the reference, achieving maximum savings of 40 % at the set-point of 22ºC and in continuous operation. However, savings were dependant on operation schedules and set-point temperatures.

Furthermore, the tests showed the peak load shifting capacity of the radiant wall, through an active use of its thermal mass. The embedded pipes can be used to store heat in the wall which is then slowly released. This asset also implies having a slow response time, however, the radiant wall can lead to further energy use reduction by operating in the periods when the heating system works on more efficient conditions. Even more, the heating can be shifted to low energy cost periods, adding economic savings to the already reduced energy use.

Despite the advantages observed in the experimentation, the results highlight the importance of using an optimized controller, which could supply the adequate heat pulse according to estimated outdoor conditions and expected internal loads.

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